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## **ADVANCING COMPRESSOR DYNAMIC SIMULATION FIDELITY WITH USE OF FIELD DATA**

### **George Talabisco**

Principal Engineer  
The Dresser-Rand business (Part of Siemens Power & Gas Division)  
Olean, New York, USA

### **Jon Paul Van Buskirk**

Senior Process Engineer  
The Dresser-Rand business (Part of Siemens Power & Gas Division)  
Houston, Texas, USA



George Talabisco is an Engineering Fellow for the Dresser-Rand business, part of Siemens Power & Gas based in Olean, NY, USA. He holds a Bachelor of Science degree in Electrical Design Engineering Technology from Penn State University. George began his career with Ingersoll-Rand 41 years ago as a Control System Design Engineer in Phillipsburg New Jersey. He relocated to Olean when the Dresser-Rand joint venture formed in 1987. George has spent most of his career specializing in controls and process engineering and enjoys helping clients solve their design and operational challenges. He has been active in ASME, has authored or co-authored several technical papers, and has two US patents pertaining to gas turbine operation and controls. George is the co-founder of the Dynamic Process Simulation team and continues to lead this Olean based Team. He is considered a resident expert in surge and process control design, package design, gas seal systems, and electric wiring for hazardous locations. George spent much of his early career at client sites and was involved in hands-on commissioning and troubleshooting. He may be reached at [george.talabisco@siemens.com](mailto:george.talabisco@siemens.com)



Jon Paul Van Buskirk, MSChE, is a Sr. Process Simulation Engineer with the Dresser-Rand business, part of Siemens Power & Gas based in Houston TX, USA. He holds a Master of Science degree in Chemical Engineering from Texas A&M University Kingsville. Jon has experience in various process chemical industries including refineries, petrochemical and the polymer industries over the past 22 years and joined Dresser-Rand in 2007. Past employment includes Aspen Technology, Inc., Quality Monitoring & Control, Lockheed Martin / Jacobs Engineering Group at NASA Johnson Space Center, and Dresser-Rand to present. Jon actively supports the Dynamic Process Simulation team purposed to evaluate, optimize, and maintain compressor systems with associated hardware. He may be reached at [jvanbuskirk@siemens.com](mailto:jvanbuskirk@siemens.com).

### **ABSTRACT**

The ability of a dynamic simulation model for a centrifugal compressor train to accurately predict the compressor operating point and performance behavior with rapid speed decay during an emergency shutdown event is a complex task. The use of actual operating data to fine tune the transient behavior of a centrifugal compressor train simulation model during an emergency shutdown event is presented. Some of the important points required to obtain a higher degree of accuracy in a compressor dynamic simulation model are also presented.

### **INTRODUCTION**

Dynamic simulation of centrifugal compressor packages can be performed by a number of different entities including engineering design contractors, equipment end users, original equipment manufacturers, education institutions and independent third party companies or consultants who specialize in dynamic simulation studies. Each of these entities has their own unique methodologies and tools used to model a centrifugal compressor package.

This discussion is intended to provide a method of improving the accuracy of the centrifugal compressor system simulation model results including the analysis and use of actual operation data for the system being modeled.



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### **DEFINITION OF EQUIPMENT TO BE MODELED**

A simulation study of a centrifugal compression package consists of the rotating equipment including the compressor, driver, connected process equipment and associated piping and valves between a defined inlet boundary and discharge boundary in a compression process. The compression train options may consist of a single stage centrifugal compressor or multiple stages of compression connected in series with individual or combined recycle loops. The compressor could also be a refrigeration service train some may include up to four side stream process feed.

The driver may be a gas turbine, steam turbine, fixed speed motor or variable speed motor. Sometimes fixed speed gearboxes or variable speed gearboxes are also utilized.

This analysis will first review a single compressor train. The compressor train consists of a variable frequency drive motor, speed increasing gearbox driving a back-to-back compressor train with a close coupled crossover pipe between the first section discharge flange and the second section suction flange. The train has a single overall recycle loop so the compressor is treated as a single section compressor. The compressor has an operating range of ~10,000 to 12,000 RPM and a load of ~25,000 HP (18.6 MW) with suction and discharge pressure of ~875 PSIA(60.3 BARA) and ~3400 PSIA(234.5 BARA) respectively. The location is offshore in the North Sea.



**Figure 1** a typical compressor train for this type of application preparing for shipment.

### **OBJECTIVES OF THE STUDY**

There can be many scenarios to analyze based on project specific requirements. Some of the more typical simulation scenarios required of a study include:

- Determine if the compressor train enters into the surge region during an Emergency Shutdown ( ESD) trip from design point.
- Identify if the compressor train enters into the surge region during an ESD trip, from an operating point on or near the surge control line.

- When and if the compressor train enters into the surge region after compressor ESD trip event, how quickly after trip initiation does the compressor enter into the surge region.



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Identify through simulation whether the only solution to preventing compressor surge on trip is by adding a time delay. A time delay on trip may not prevent surge for every trip event including ESD events that require immediate shutdown like main drive motor fault or fire & gas detection

Determine if entrance into surge can be mitigated by changing anti surge valve dead time, overall trip response time, valve characteristic or valve overall flow capacity (Cv). It could be a combination of some or all of these characteristics. (White and Kurz 2006; Samurin and Talabisco 2007)

In some cases installing an additional valve in parallel with the anti-surge valve or a separate short circuit hot gas bypass valve can prevent the compressor from entering surge during an ESD trip event. (White and Kurz 2006; Samurin and Talabisco 2007)

Identify if the compressor recycle system can protect the compressor from going into surge during an upstream or downstream blockage event (i.e. a block valve closure)

Identify by simulation if the compressor train can exceed the relief valve or high pressure trip setting during a blockage event

Use simulation to determine whether the compressor drive train power limit can be exceeded during a full open anti-surge valve event (e.g. operator intervention or valve fail open scenario)

Investigate the scenario when an anti-surge valve full open event would allow the compressor to operate deep in overload

Confirm correct response of system & controls to transients when multiple units are operating in a series or parallel configuration

### **COLLECTION OF NECESSARY DATA AND DEFINE OR CONFIRM BOUNDARY CONDITIONS**

The amount of data required to perform an accurate simulation study is substantial and sometimes not all information is available or finalized at the time a simulation study is executed. This can lead to additional assumptions required with associated risks of inaccuracy that can impact the predictions of a dynamic simulation model. A list of data required to perform a simulation study is generated and provided to the client, engineering contractor and sub-suppliers to collect and provide the necessary data. This list including usual source of the data can include the following:

- Process flow diagram(s) and heat and mass balances for the design condition. (client/contractor)

- Process and instrumentation diagram(s). (client/contractor)

- Volumes of the piping, associated vessels and coolers. The volumes are to be for the individual inlet and discharge components of the compressor (client/contractor). Note the volume boundaries are usually the inlet block/check valve, anti-surge/recycle valves, discharge block/check valve. These boundaries need to be confirmed.

- Piping approximate pressure losses, pipe size, schedule and pipe system volumes (client/contractor)

- Heat exchangers: Volumes, duties, pressure losses, pressure rating and coolant supply temperature. (client/contractor)

- Vessels: normal gas volumes and any appropriate pressure losses, max pressure rating (client/contractor)

- Anti-surge control valve data sheets showing: (vendor/client )

  - Valve sizing pressures, flows, flow conditions and maximum capacity (Cv)

  - Valve body style type e.g. globe, ball, butterfly

  - Valve characteristic as in linear, equal percentage or quick opening

  - Valve stroke time from full closed to full open and full open to full closed

  - Valve failure position (open or closed)

- Block isolation and inlet throttle valve data sheets showing: (client/contractor)

  - Valve body size and maximum capacity (Cv)

  - Valve body style type e.g. globe, ball, butterfly

  - Valve characteristic as in linear, equal percentage or quick opening

  - Valve stroke time from full closed to full open and from full open to full closed

  - Valve failure position (open or closed)

- Blowdown valves data sheets showing: (client/contractor)

  - Valve body size and maximum capacity (Cv)

  - Valve orifice sizing

  - Valve body style type e.g. globe, ball, butterfly

  - Valve characteristic as in linear, equal percentage or quick opening



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Valve stroke time from full closed to full open and from full open to full closed  
Control philosophy document defining mode(s) of control (client/vendor/contractor)  
Start-up and shutdown procedures in written form or flow-chart (client/vendor/contractor)  
Relief valve data sheets (client/contractor)  
Process data at battery limits of study including: pressure, temperature, flow, gas analysis (client/contractor)  
Electrical system start up voltage dip and recovery characteristic (client/contractor)  
Gearbox variable speed- torque characteristics if included (Vendor)  
Motor driver start-up and shutdown characteristics (speed torque data at identified voltage dip if required (vendor)  
Electrical system voltage dip and recovery for motor start-up are required for main drive motor (client/contractor)  
Time limit of motor during start-up sequence if motor driver (Vendor)  
Steam or gas turbine driver start up and shutdown characteristics including speed-torque characteristic (vendor)  
Train rotating inertia ( $WR^2$ ) of all components, driver, gearbox, compressor, couplings, etc. (vendor)  
Compressor maps based on operating conditions (compressor vendor)

The simulation model is constructed and populated with data for steady state validation to comply with the design basis before any transient scenarios are exercised. Steady state model validation is an important step prior to exercising the simulation model scenarios.

#### **MODEL DESCRIPTION**

Figure 6 in the Appendix shows the simulation model flow sheet for the VFD motor driven compressor train. This is a two section compressor with crossover pipe so it is modelled as one compressor section with a single overall recycle loop. This model consists of a feed stream (right arrow) connected to an inlet block & check valve combination. The outlet of the valve feeds a stream mixer where the recycle stream and Hot Gas bypass stream connect with the incoming feed stream. The outlet of the stream mixer then feeds a suction separator. The suction separator connects to the suction volume which in turn connects to the compressor inlet. The compressor outlet stream is connected to a stream splitter with the main stream of hot gas continuing on to the hot discharge volume. The split stream is connected to a Hot Gas Bypass Valve which recirculates hot gas back to the compressor suction only during a trip event. The hot gas discharge volume is connected to a hot discharge check valve. This hot check valve is used to isolate a smaller working volume upon compressor trip event. This volume is between the compressor discharge flange, the hot discharge check valve and the hot gas bypass valve. The outlet of the hot check valve connects to the discharge gas cooler. After the gas passes thru the discharge cooler the gas enters another stream splitter. The main stream of cooled gas is connected to the discharge volume block. The split stream is connected to the anti- surge recycle valve which returns cooled recycle gas back to the compressor suction upstream of the suction scrubber where it mixes with the incoming gas from the main feed source and the hot gas bypass return during a trip event. The gas flow from the discharge volume block connects to the discharge check valve & block valve combination. After the gas passes thru the discharge block and check valve combination it exits the system. The suction volume was  $\sim 1050 \text{ FT}^3$  ( $29.7 \text{ M}^3$ ). The hot discharge volume was  $\sim 90 \text{ FT}^3$  ( $2.5 \text{ M}^3$ ) and the cooled discharge volume was  $\sim 180 \text{ FT}^3$  ( $5.1 \text{ M}^3$ )

Figure 7 in the Appendix is the flow sheet legend for the functional blocks used within the simulation modelling software for this compressor train.

#### **DEFINE ASSUMPTIONS TO BE USED AND VALIDATION**

For items that cannot be fully identified, assumptions are made based on engineering judgment and field experience. The assumptions used in modeling are important to obtain accurate results and can sometimes lead to different results if not accurate. Below is a typical list of assumptions that is often used:

- Compressor design certified operating case used as starting point.
- Design point at rated design speed basis for evaluation
- Definition of gas properties equations of state used in simulation model
- System volumes based on client supplied data, as given in the appendices.
- The pipe and vessel volumes are combined as dynamic volumes located at the inlet and discharge of each compressor section.
- All pressure losses are taken at valves. No pressure losses for pipes, vessels or coolers are taken separately except where special circumstances apply (i.e. flow limiting condition)
- Client pressure loss data is provided and used in model.
- Anti-surge valve vendor data as provided by the vendor and used in simulation model including valve capacity and characteristic (  $C_v$  versus percent open)





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The emergency shutdown, trip open stroke time of 0.5 seconds with a 100 millisecond delay time as specified for the compressor anti-surge valve (ASV), per vendor supplied datasheets

A normal positioner controlled stroke time of 1.0 seconds for the anti-surge valve per vendor supplied Data sheets.

The hot-gas bypass valve (HGBV) Cv requirement for each compressor section are identified:

HGBV1 Cv = 500 -550(Quick-Open) for Stage 1 ESD/trip bypass valves (typical value)

The stroking time from closed to opened is 500 milliseconds for each HGBV plus a 100 milliseconds delay time to begin to open as identified from vendor data sheet

All valves are specified as linear characteristic valves unless otherwise noted

Rotating inertia ( $WR^2$ ) is referenced to train compressor speed.  $WR^2 = 280,000 \text{ lb-in}^2$  (unique to each equipment train) including rotating Inertia ( $WR^2$ ) values as provided by driver, gearbox, compressor and coupling vendors

Mechanical losses are taken for bearing, seals and gearbox

Compressor model is within +/- 2% of the predicted or as-tested performance

OEM compressor modeling is used for performance evaluation

Delay times including shutdown system update time, driver trip system delay time (fuel, steam or electric power shutoff time and power decay)

Driver residual power production decay function as provided from supplier

Windage losses for driver as provided from supplier

Driver downstream stored energy capability as provided from supplier

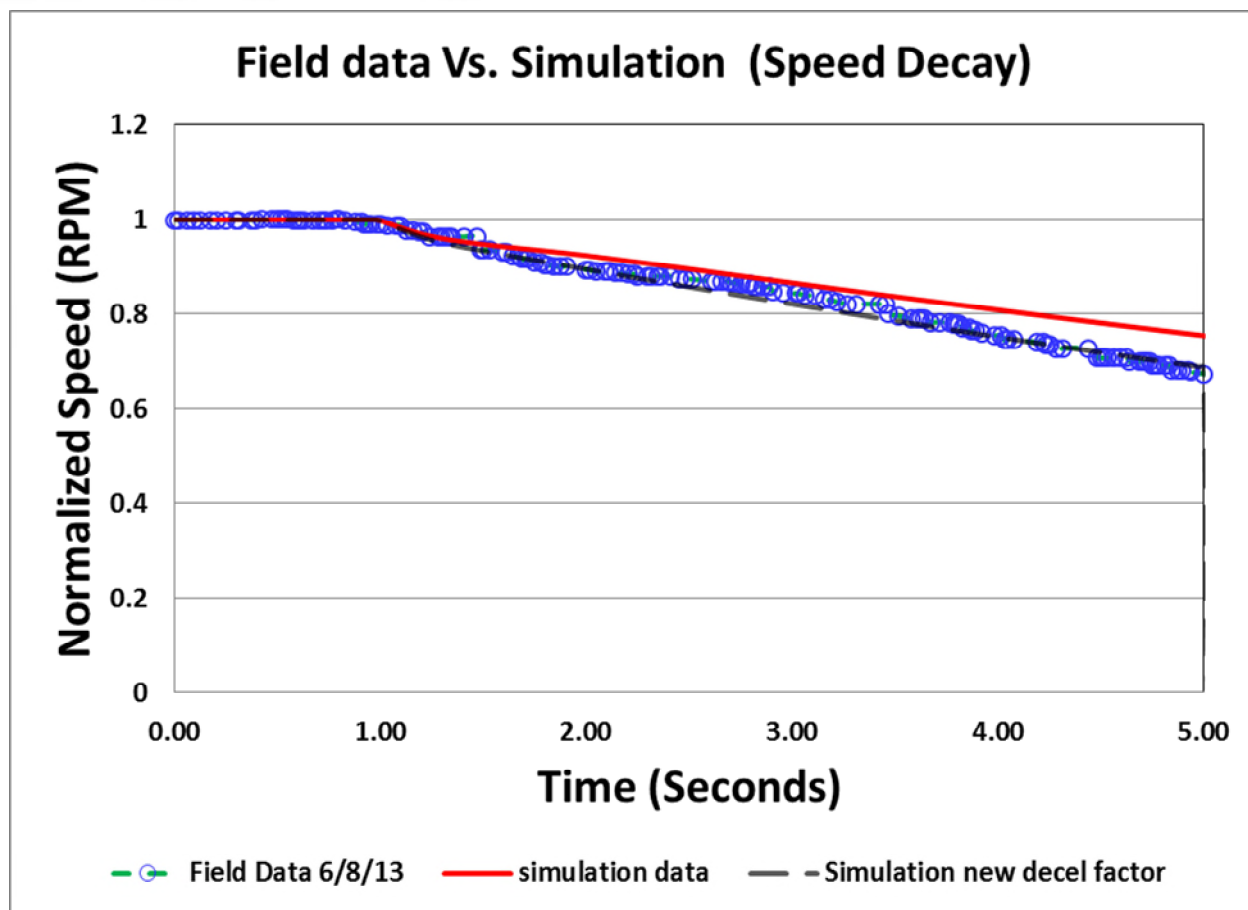
Boundary conditions are needed to accurately model the simulation model response. This includes pressure, temperature, flow, gas constituents and also upstream or downstream control actions that can interact with the compression system. This is important to understand and utilize in both single and multiple compression train models.

**RUN SCENARIOS AND COMPARE RESULTS TO MEASURED FIELD DATA AND FILTER OUT ANY OUTLIERS**

Typical ESD trip run with speed versus time, head versus flow and pressure versus time are shown for the model described above and discussed below in Figures 2, 3 and 4 respectively.



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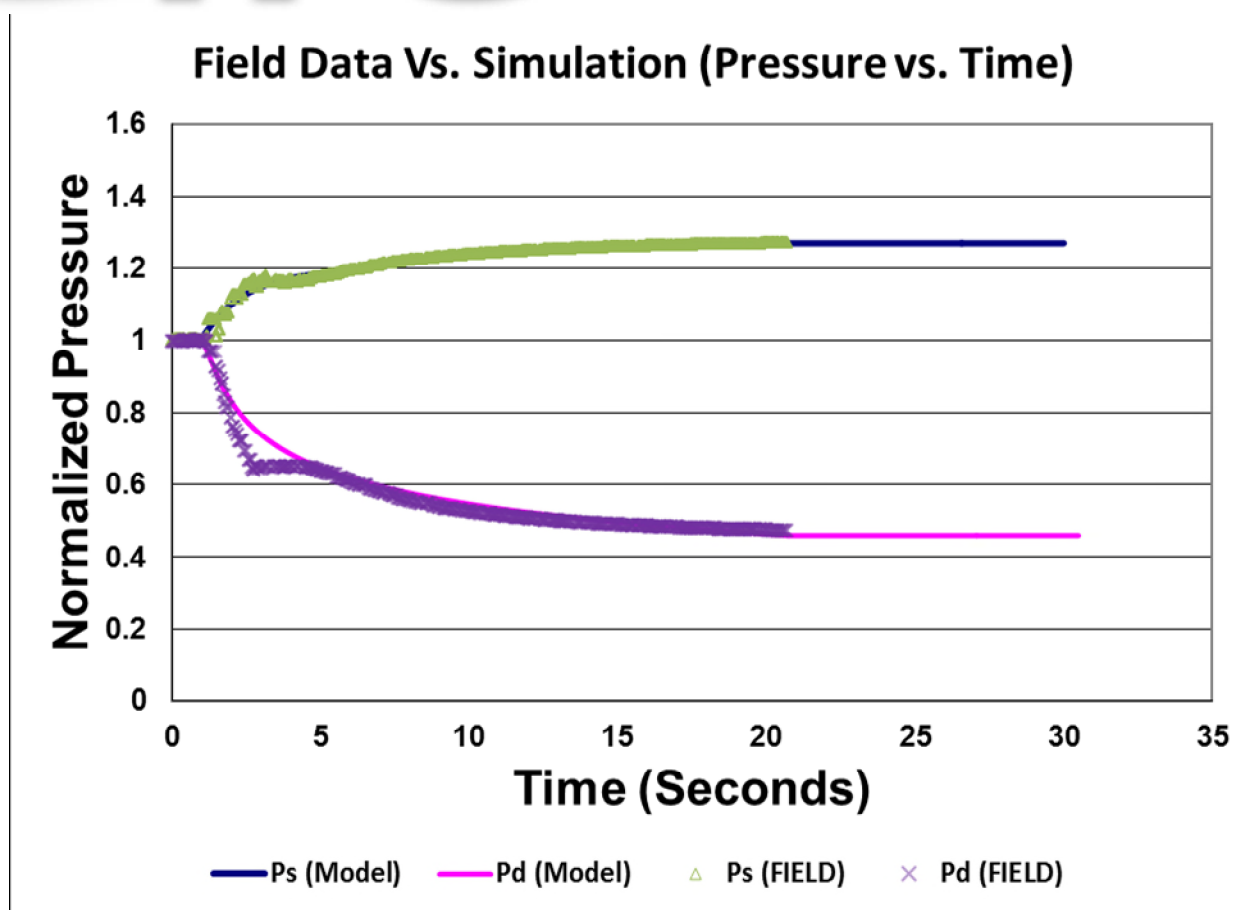


**Figure 2 Normalized Speed versus Time comparison of field data to simulation model**

Note that the model was run at steady state for 1 second prior to initiating the ESD trip event. The speed deceleration for the updated simulation model above shows good correlation with field data during the first two seconds of deceleration after initiation of ESD trip event. The original simulation model predicted a somewhat slower deceleration rate while the actual field data showed increasing divergence from the simulation model around two seconds into the ESD trip event. Since the first two seconds into a transient event are important for compressor surge protection, it is better to have accurate modeling during the initial time period after trip is initiated. If a simulation model does not accurately predict speed decay then the rest of the simulation results can be skewed and not provide an accurate prediction of what is actually happening with the equipment. It is important to know the point in time during the trip sequence the simulation model is deviating from the field measured data. Modifications to the simulation model deceleration calculation by adjusting the horsepower losses was required for time greater than 2 seconds after ESD trip initiation based on the actual field data. Some additional discussion on resolving simulation model deviations from measured data is discussed later in this paper.



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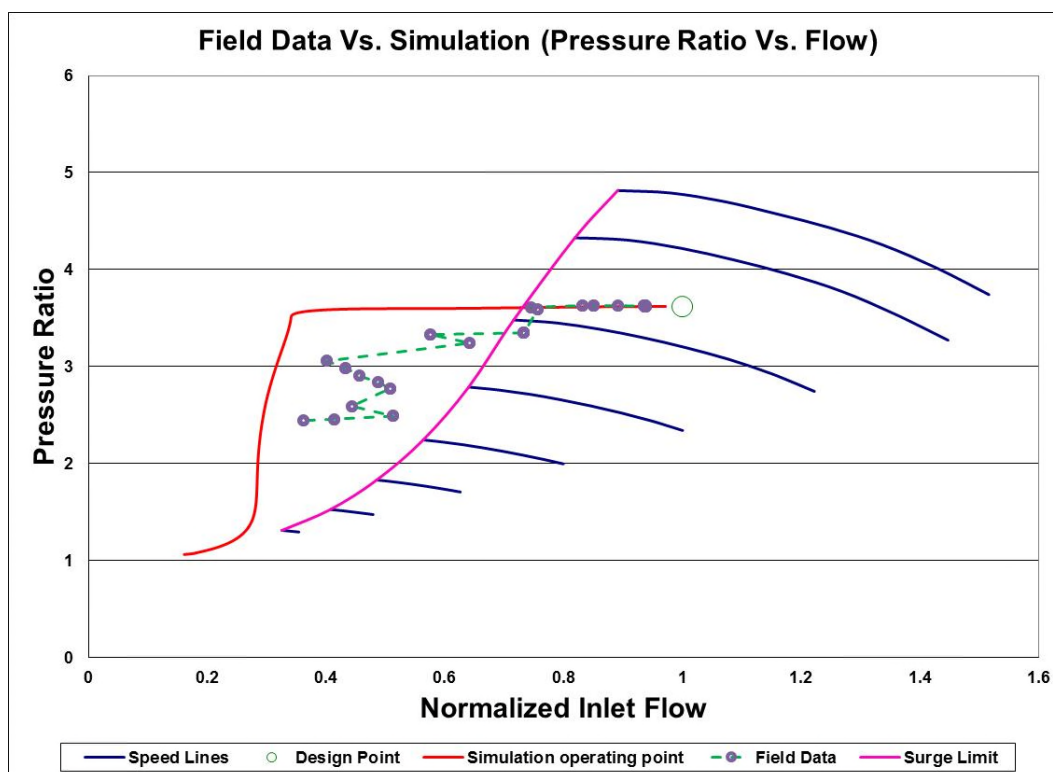


**Figure 3 Normalized Pressure versus Time comparison of field data to simulation model**

Figure 3 shows a comparison of pressure readings against model predictions. The normalized suction and discharge pressures in the simulation model match the field data reasonably well during most of the shutdown event. There is a point around 1 to 3 seconds where discharge pressure field data shows deviation from predictions. This is due to compressor surge event (as seen in figure 4) causing process instabilities. Reasoning a surge event occurs in an area of unpredictability (compressor surge region) where compressor simulation model is not accurately predicting compressor performance. Many dynamic process simulation tools have a number of adjustment parameters to match field data very accurately including compressor performance, Equation of State (EoS), process volumes, initial process conditions, etc. Modeling adjustments are sometimes required especially during times of unstable process operating conditions. The primary goal of the simulation is for identifying unstable operation and taking corrective actions to prevent the undesirable operation such as compressor surge events. Knowing an undesirable event may occur allows development of corrective actions to prevent or mitigate the events from occurring. The simulation model can then be used to identify possible and feasible solutions in order to help assure reliable long term compressor operation.



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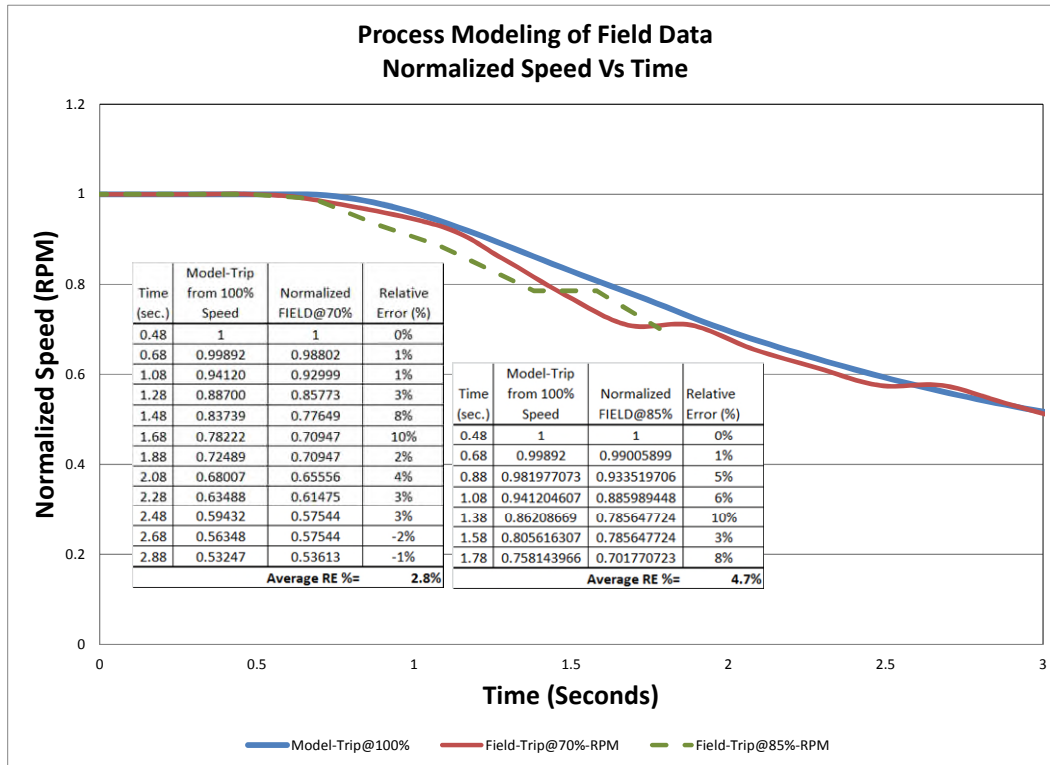
**Figure 4 Pressure Ratio versus Normalized Flow comparison of field data to simulation model**

Figure 4 shows a normalized flow versus pressure ratio graph of an ESD trip event. Note that the model has good initial prediction (red line on graph) indicating an excursion into surge but once the compressor operating point is in the surge region (located to the left of the surge limit line) the model is not able to predict compressor performance accurately. The simulation model uses the pressures in the suction volume and discharge volume to calculate the pressure ratio. This ratio along with the calculated speed change within the simulation model determines a flow. With the flow and pressure ratio the simulation model now has the data to determine the compressor operating point. This calculation does not change regardless of whether the compressor is operating within the compressor performance envelope or to the left of the surge limit line. The compressor simulation as modeled did not calculate flow reversal when in the surge region and would require a lot more actual field operating data than was available to accurately assess and model the compressor response. (Moore et. al. 2008; Moore et. al. 2009) The field data shows some flow loss then some limited flow increase (or recovery) but not enough to achieve steady operation to the right of the surge limit. The sample rate of the field data is slower than the simulation time step that the calculations are performed within the simulation model. This slower sample rate does not necessarily provide a complete picture of the actual event but does show a back and forth cyclic nature of compressor flow when operating in the surge region. This cyclic increase and decrease of flow (instability) within the surge region shows the compressor cannot produce enough head to satisfy the system head required and is typical of a compressor surge event. The head required is based on the pressure difference between the suction volume and the discharge volume. Predicting compressor performance within the surge region is a complex and debatable subject that is beyond the scope of this discussion. As a side note there is published literature & work in the field of compressor performance while operating in the surge region that could be investigated to improve accuracy of compressor performance prediction while operating in the surge region. Since the compressor operating point did cross into the surge region on an ESD trip, this identified a recommendation for system modification including addition of a Hot Gas Bypass Valve and Hot Gas Check Valve to mitigate and possibly prevent compressor surge on a trip event. (Botros et. al. 2015; Patel et. al. 2007; White and Kurz 2006)





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**Figure 5 Normalized Speed versus Time plot comparison of field data to simulation model**

The above normalized deceleration curve (Figure 5) is from a different application consisting of a two shaft GT (gas turbine) driver, speed increasing gearbox and a single compressor case with two separate & independent compressor sections. The application is export gas service with a compressor design operating load of ~ 42,900 HP (32 MW) with a first section inlet pressure of ~ 750 PSIA (51.7 BARA) and second section discharge pressure of ~ 1600 PSIA (110.3 BARA). The client provided a limited amount of data after the equipment was installed and commissioned. Comparison of the limited field operating data to the simulation model data shows there is good relative error correlation between the simulation model and actual field data for the 70 % speed trip during the first 0.8 seconds after trip initiation and on an overall average relative error basis (2.8%). The 85 % speed field trip data shows an offset which indicated a larger deviation during a compressor surge event that was experienced during the actual shutdown. The relative errors against the normalized speed decay trip model from 100% speed and the client field data (initiated from 70 & 85% speed) are within expected reasonable accuracy considering that the process is experiencing an upset during an actual surge event. The dynamic process simulation model was used along with other field data to confirm a compressor surge occurrence at the site location and became instrumental in determining a solution to eliminate the surge occurrence. (Patel et. al. 2007)



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## **DEVELOP MODIFICATIONS REQUIRED TO OBTAIN MODEL CORRELATION TO ACTUAL EQUIPMENT BEHAVIOR**

Modifying the deceleration rate to match actual measured values requires comparison of the two plots versus time and evaluating if changes are required. The modification must be able to cover the first few seconds after trip and usually fall into the following categories:

- All the delays in control system and component response must be properly accounted for

- The delay time in depleting all the energy provided to the driver including fuel isolation system shutoff time and residual energy of fuel downstream of shutoff valve

- The power producing capability of the driver (gas or steam turbine) as it coasts down to rest since it is still producing some torque.

All of these parameters and others listed in the assumptions must be accurately accounted for to obtain a realistic response from a dynamic simulation model. Changes to system time delays, power decay function versus speed and losses versus speed are not always well defined and sometimes need to be modified to obtain a more accurate simulation model.

Sometimes deviation from measured field data is the result of an event occurring in the real system versus how it was modeled in the simulation. As an example this deviation could be a logic event causing additional valve action to occur that was not known hence modeled inaccurately. The deviation could also mean there could be a choked flow condition somewhere in the system or a number of other possibilities. In this situation, close coordination with equipment operators and process designers to fully understand the process, equipment operating sequence and operator action is required to confirm a simulation is accurately portraying system response.

## **DISCUSSION OF RESULTS AND PITFALLS**

Speed decay specifically during the first few seconds after an ESD trip event is an important parameter. (Botros et.al. 2015; Moore et. al. 2008; Moore et. al. 2009; Samurin and Talabisco 2007; White and Kurz 2006)

System time lags are needed to develop an accurate simulation model. The simulation model has to consider the following:

- All the delays in control system and component response must be properly accounted for

- The delay time in depleting all the energy provided to the driver including fuel isolation system shutoff time and residual energy of fuel downstream of shutoff valve

The difference in torque is calculated for each time step within the simulation from the power difference when the driver is tripped by the formula that calculates the power from equation 1

$$\Delta HP = HP_{driver} - GHP_{comp} - HP_{losses} \quad (1)$$

For an electric motor the driver power becomes zero when the motor is disconnected from the power source. For a gas turbine or steam turbine driver there is some residual power developed after the driver is trip is initiated. This residual power initially contributes some power immediately after an ESD event is initiated. The torque difference is derived from the power difference during deceleration.

The torque is related to power delta and speed at that specific time by:

$$HP = \frac{Torque \times N(t)}{5250} \quad (2)$$

Then by rearrangement of equation 2 to obtain the torque difference in equation 3



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$$\Delta Torque = \frac{\Delta HP \times 5250}{N(t)} \quad (3)$$

The change in speed with respect to time is then calculated by equation 4

$$\frac{dN}{dt} = \frac{\Delta torque \times \Delta time}{Wr^2 \times N(t)} \quad (4)$$

With the above calculations the rate of speed decay is calculated for each time step within the simulation model. At each time step the new speed value is then used to re-calculate the next set of values as the simulation progresses through compressor coast down.

## CONCLUSIONS

Dynamic Simulation has been in use by the oil and gas industry for many years. The ability to validate a simulation model with actual field data is not always available at the time of study execution, particularly when used during the design stage of a new installation. The benefit of having experience with previous study work and field data from similar equipment to more accurately predict how a new installation will behave during high speed transient events is valuable. Personnel involved in dynamic simulation have to be able to constantly adapt and learn from our failures and success stories. The personnel involved in performing dynamic simulation must keep in mind that very few systems and their respective dynamic response are identical. Small differences including process volumes, equipment layout, valve characteristics, different generation design of control components and control system response time all factor into the overall system response. It is important to not assume systems will behave like they did in the past during earlier similar designs or how it was predicted in the past. Simulation personnel should always try to validate the simulation work with solid and reliable field data. This provides a continuous stream of validation and improvement to prediction accuracy. This is an ongoing method to assure the simulations that are developed, produce reliable results from simulation studies.

## NOMENCLATURE

Crossover Pipe	= Piping directly connecting a compressor section discharge nozzle to the next section suction nozzle of a compressor
$C_v$	= Valve Sizing Coefficient where a $C_v=1$ is the valve capacity required to pass one U.S. GPM of water at 60 degrees F. with a 1 lb. pressure drop
ESD	= Emergency Shutdown
GT	= Gas Turbine
Flow Mass	= Flow (lb. / Minute)
Flow Volumetric	= Flow in Actual Cubic Feet per Minute (ACFM)
Head	= Polytropic (Foot-lb. / lb. Mass)
HP losses	= losses due to friction windage etc.
MW	= Mole Weight
N	= Speed in revolutions per minute (RPM)
Ps	= Suction Pressure (PSIA)
Pd	= Discharge Pressure (PSIA)
Temperature	= Degrees Fahrenheit
t	= time (based on calculation time step)
VSD	= Variable Speed Drive for Main Drive Motor
$Wr^2$	= Rotating Inertia (lb-in <sup>2</sup> )



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## APPENDIX

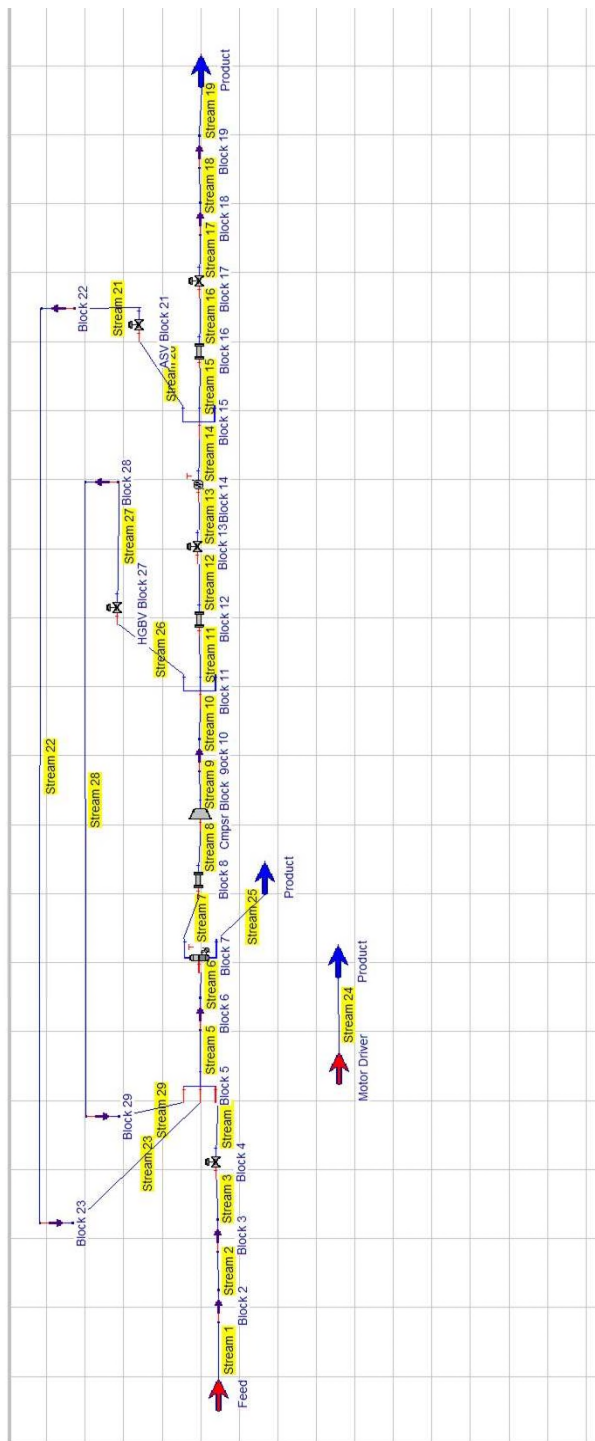
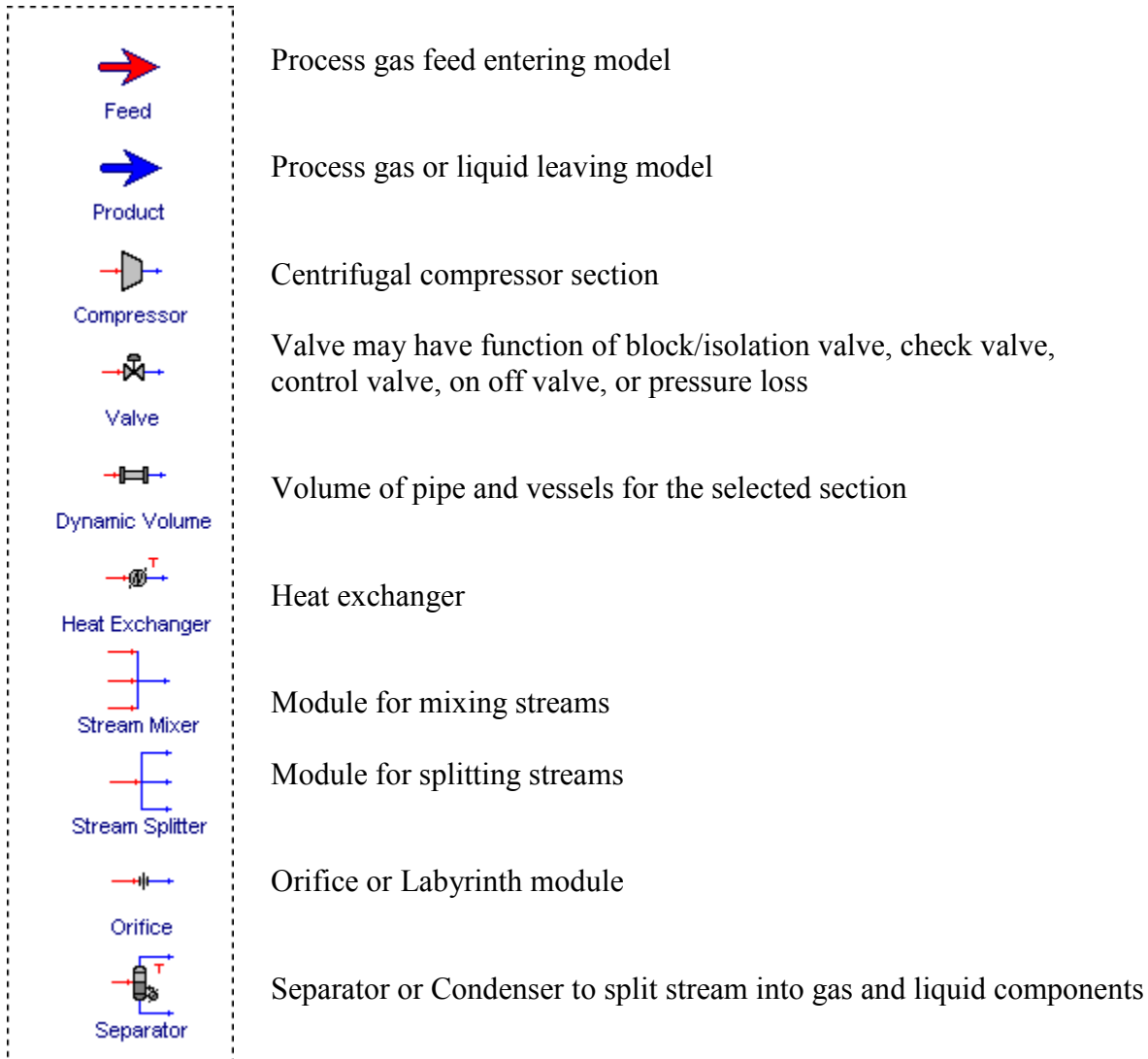


Figure 6 - Simulation model flow sheet for a compressor utilizing a single overall recycle loop with variable speed drive electric motor (Franks 1972)



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**Figure 7- Flow sheet symbols legend**  
 (Franks 1972)





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